Investigation into a stepped piston engine solution for automotive range-extender and hybrid electric vehicles to meet future green transportation objectives

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Abstract: Securing the objectives for future high efficiency low CO_2 vehicles is a key target for automotive manufacturers. This paper considers a high durability two-stroke cycle engine in terms of performance and computational modelling of emission characteristics for automotive range-extender or hybrid electric vehicle power plant application. The engine uses novel segregated pump charging via the application of stepped pistons and the study compares the engine characteristics with a comparable four stroke cycle engine of similar expected power output (60+ kW per litre). In the interests of cost minimisation both engines are limited to parallel two cylinder in-line configurations with the intention of still being able to achieve acceptably low noise, vibration and harshness (NVH) characteristics. In order to achieve low engine exhaust emissions computational modelling of direct injection is considered for the stepped piston engine. Significant NOx emission reduction of between 31% and 55% is observed.

Keywords: Hybrid electric vehicle, range extender electric vehicle, stepped piston engine, two-stroke cycle engine, four-stroke cycle engine, NVH, engine modelling/simulation

1 INTRODUCTION

Small internal combustion engines with fewer than four cylinders have been considered by a number of researchers as suitable power plants as the prime mover to provide on board power generation to effectively address the range limitation of electric vehicles [1][2]. The Range Extender Electric Vehicle (RE-EV) has therefore been conceived to address this need and notable examples such as the GM Volt [3] have shown successful application. The GM Volt however does use an inline 4-cylinder power plant. The cost implications of such engines when considering the powertrain duplication required for RE-EVs and Hybrid Electric Vehicles (HEVs) present serious challenges to engineers in attempts to introduce viable solutions that are cost effective to produce. Whilst such vehicles readily address CO_2 reduction strategies to meet current and future government legislation they are often produced with minimal profit margin in order to address these needs and still try to present an attractive investment to the prospective vehicle owner. Naturally if production costs can be reduced then RE-EVs and HEVs become more viable in an already competitive market.

Reducing the number of cylinders selected for the IC engine solution provides a significant cost reduction strategy. Two cylinder possibilities have been researched such as four stroke units by the work of Bassett *et al* [4] and some examples of two stroke cycle engines have been considered such as the work of Stan *et al* [5] and Mattarelli *et al* [2][6]. Whichever fundamental choice between two or four stroke cycle is taken for the power plant stringent emissions targets must be meet. Direct fuel injection is a requirement for the two stroke engine. The prior work of Duret *et al* [7], Schlunke [8], Osborne *et al* [9] and Turner *et al* [10] has demonstrated significant emissions reduction for two stroke engines. The published work of Blundell *et al* [11] has demonstrated extremely low NOx emission pushing the limits of being able to record such low 3ppm concentrations whilst operating at 1.5 bar IMEP at 2000 rpm operating conditions.

Reducing cylinder numbers effectively contains the cost of the powertrain system however this comes at an additional penalty in terms of increased noise vibration and harshness (NVH). The two stroke engine, for the same number of cylinders, presents a significant advantage in this context. If such a low emission efficient power plant was feasible and was able to offer high levels of in service durability, then a system capable of meeting modern vehicle levels of refinement would be an attractive prospect. The lower engine mass inherent in many two stroke designs when compared to comparable four stroke systems provides a further benefit.

Two stroke engines due to their inherent simplicity in the region of cylinder head construction offer simpler solutions for potential variable compression ratio (VCR) operation as discussed by Turner *et al* [10], Hooper *et al* [12] and Stone [13]. In simple RE-EV systems this may be of limited use as the engine would typically be required to operate at a single load condition or a very limited range of loads. However for greater sophistication the relative simplicity of VCR could offer further CO_2 reduction over a wider range of operating conditions.

The study presented in this paper considers a two stroke cycle engine using segregated scavenging. The engine uses stepped or two diameter pistons. This technology has proven to be able to achieve considerably higher durability than can be achieved with conventional crankcase scavenged two stroke cycle engines. The study compares attributes of the engine type with a hypothetical four stroke engine of comparable output. In order to present solutions trying to address the low production cost objectives both engines are limited to two-cylinder arrangements whilst still aiming to meet the additional low NVH objective.

2 CONVENTIONAL CRANKCASE SCAVENGED TWO-STROKE CYCLE ENGINES

Conventional two stroke cycle engines offer a simple low cost charge scavenging system by employing the crankcase as a pumping means. Unfortunately as higher levels of thermal efficiency and reduced exhaust emissions are demanded the short comings of such an approach become increasingly evident. The fact that the lubrication system and intake air transfer systems are effectively combined within the crankcase means that there is inevitable cross contamination of the charge air with lubricating oil. Effective means have been developed to provide greater levels of control of lubricating oil flow and recirculation of excess lubricant as demonstrated by the research of Shawcross *et al* [14] however with ever more stringent environmental legislation this is a limited solution for future reductions in HC emission and leads to a conflict of interest within the conventional two stroke engine. The application of direct injection to two stroke engines has demonstrated very low emission levels however the thermal loading of the piston increases as a consequence and the need for higher levels of cooling and lubrication are required rather than a reduction in oil consumption to counter HC emissions from the lubricating oil as a source. If the crankcase can be isolated from the air charge transfer system, then significantly lower levels of oil consumption and greater levels of durability can be achieved.

3 STEPPED PISTON ENGINES

The principle of operation of the stepped piston engine is shown for reference in Fig 1.

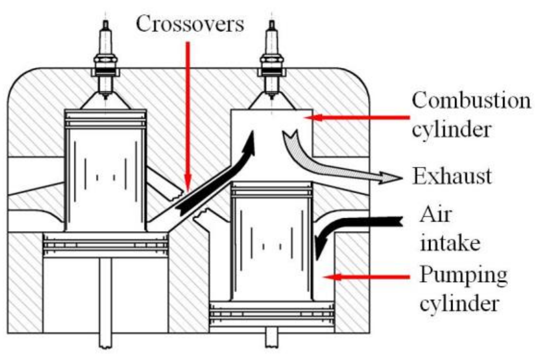

Fig.1. Simplified schematic of the SPX stepped piston crossover system
(image courtesy of Bernard Hooper Engineering Ltd)

The incoming fresh air charge is drawn into the pumping cylinders of the engine via the air intake. This is normally controlled by a reed valve or other valve means thereby preventing reverse flow after bottom dead centre. As the piston ascends the trapped charge is transferred via the crossover system into the relevant paired power or combustion cylinder. Multiple transfer ports (not shown in Fig 1) are arranged to employ the well-established Schnürle loop scavenging principle. Combustion then takes place in the upper cylinder and the combustion products are expelled via the exhaust port.

The pumping cylinder effectively replaces the crankcase scavenging process of the conventional two stroke engine and therefore isolates the lubrication method from the air delivery process. Compression rings are applied to the power piston and a compression and oil control ring are positioned on the pumping piston typically as shown in Fig 2.

Bearing of the two piston skirts is provided by both the combustion and pumping cylinders leading to low ring wear characteristics and reduced noise emanating from the piston as a source due to the greater level of piston to bore guidance as discussed by Hooper [15]. Positioning the gudgeon pin towards the low temperature zone of the piston has so far resulted in a zero ovality requirement.

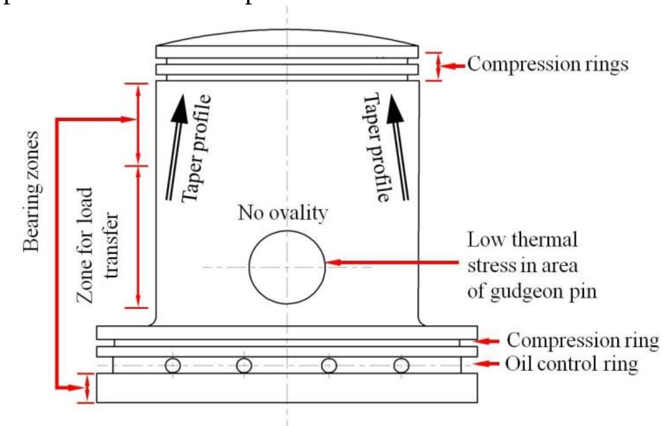

Fig.2. Piston design strategy
(image courtesy of Bernard Hooper Engineering Ltd)

Separation of the charge scavenging and lubrication processes allows critical advantages in terms of higher levels of durability than are achievable with conventional crankcase scavenged two-stroke cycle engines. Not only is it possible to apply a typical four stroke re-circulatory system but when compared with a conventional crankcase scavenged two stroke engine it is relatively simple to be able to strategically lubricate key areas of the piston. If necessary dedicated lubricant jets can be arranged from the crankcase as is sometimes applied in four stroke engine practice to target key target areas. This has so far only been applied on prototype compression ignition diesel stepped piston engines and has not yet been found necessary on spark ignition engines. The SI engines normally rely on splash feed from the pressure fed crankshaft big end bearings.

Experimental testing from prototype stepped piston engines [12][16][17] has shown that the power density of the engine type lies between the conventional four-stroke and two-stroke cycle engine levels. The relative size of current stepped piston engines is similarly between the two fundamental engine types as can be seen in Fig 3.

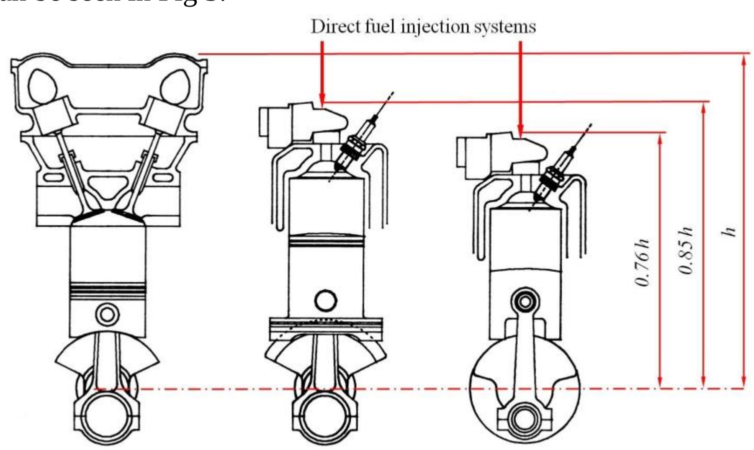

Fig.3. Identical swept volume engines (i) four-stroke DOHC, (ii) DI stepped piston and (iii) DI crankcase scavenged two-stroke cycle
(image courtesy of Bernard Hooper Engineering Ltd)

The engines shown in Fig 3 are compared in terms of identical cylinder swept volumes. Both the stepped piston and conventional two-stroke cycle engines are shown with direct fuel injection as an essential requirement for low exhaust emission. It should be pointed out that for the same given swept volume and operational speed, the DOHC four-stroke cycle engine would provide a lower power output than the conventional two-stroke cycle and stepped piston engines. To achieve an identical output level an increased swept volume would be required for the four-stroke cycle engine shown.

A compact IC engine power plant is essential for small Range Extender Electric Vehicles. Reduction of the number of cylinders is an effective means of cost limitation within the confines of achievement of acceptable levels of NVH refinement as discussed in prior publications [15][18]. Whilst at Norton Villiers, Hooper and Favill developed single cylinder 90 cm^3, 150 cm³ and 270 cm³ industrial engines and a parallel two cylinder 500 cm³ engine for a new range of Norton motorcycles as demonstrated by the Norton WULF SPX500. Each of these engines was developed in order to minimise manufacturing costs compared with conventional two and four stroke engine units produced by Norton Villiers as discussed by Hooper and Favill [19]. The latter SPX500 500 cm³ power unit was actually constrained by the need for a common crankcase for a later 750 cm³ two-cylinder engine spacing in order to achieve further cost reduction via component commonality. Unfortunately this of course means that the 500 cm³ unit is excessively large due to wider than necessary cylinder centre distance. More recently more compact engines have been developed such as the SPV580 V4-cylinder multi-fuel UAV engine [17] as shown in Fig 4. This unit design was less constrained and benefits from a more optimised cylinder centre distance.

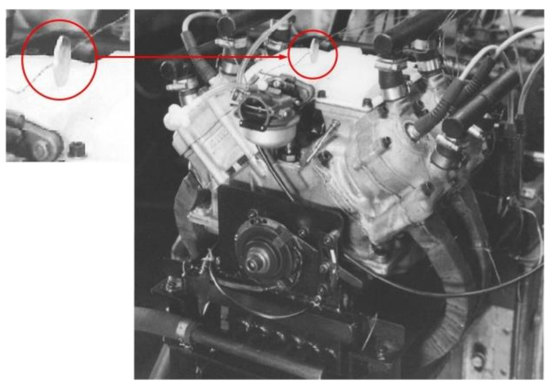

Fig.4. SPV580 stepped piston 580 cm³ 90° V-4 cylinder engine
(image courtesy of Bernard Hooper Engineering Ltd)

Close examination of Fig 4 shows a highlighted coin of approximately 30mm diameter standing on its edge on the upper surface of the engine intake manifold. The engine is operating at a simulated 22 kW 4000 rpm cruise condition and gives an indication of the low inherent vibration levels evident with a V-4 cylinder two-stroke engine configuration. However for a Range Extender a V-4 arrangement whilst exhibiting very low vibration and hence low NVH would not necessarily be best suited in terms of installation constraints for small low cost RE-EVs. An inline cylinder arrangement would be better suited to the installation and a parallel two-cylinder engine is proposed as a suitable compact power plant. A two-cylinder version of the SPV580 has been developed as a 290 cm³ engine. This power plant is shown in Fig 5 as an electrical power generator engine for marine application. The power output of the 290 cm³ would preclude use for large RE-EVs however the output of 17.7 kW is comparable with the output achieved by the Peugeot Citroen Dynavolt series hybrid vehicle [5]. The two cylinder unit is based upon the larger V-4 cylinder engine technology and provides a starting point for comparison with a similar output two-cylinder four stroke engine.

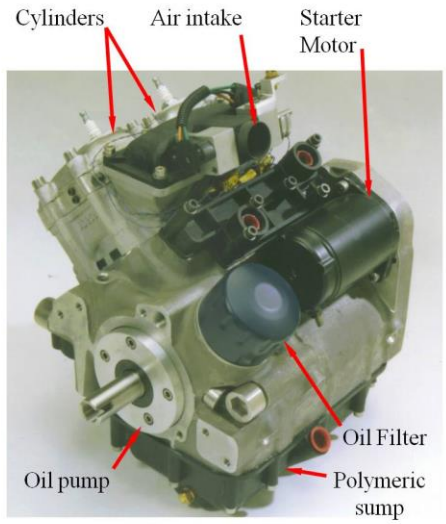

Fig.5. Stepped piston 290 cm³ 180° parallel twin cylinder engine
(image courtesy of Bernard Hooper Engineering Ltd)

Using the same cylinder centre distance as its parent V-4 cylinder engine provides minimisation of the rocking couple associated with parallel twin cylinder engines. An understandable criticism of the additional footprint required by the pumping piston diameter of the stepped piston would suggest that a greater cylinder centre distance is required compared with competing alternative power plants. However in Fig 6 a comparison of layouts for a crankcase scavenged two cylinder engine using loop scavenging and a stepped piston engine arrangement is presented.

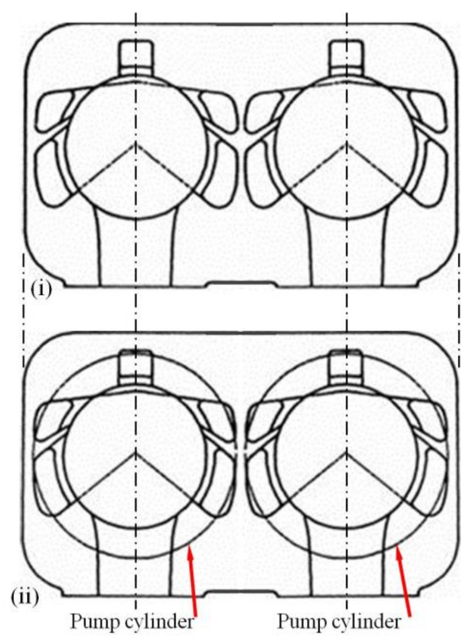

Fig.6. Comparison of porting profiles (i) crankcase scavenged two-stroke cycle and (ii) stepped piston 180° parallel twin cylinder engines
(image courtesy of Bernard Hooper Engineering Ltd)

It can be seen that no increase in cylinder centre distance is evident with the stepped piston engine when loop scavenging is applied. The centre distance for the crankcase scavenged engine could be reduced if cross scavenging was applied, however cross scavenging usually results in lower performance than the specific output achievable with loop scavenging. The rocking couple evident with two cylinder engines can be opposed by a counter balance shaft. A design scheme offering such a system added to a 290 cm³ stepped piston engine can be seen in Fig 7.

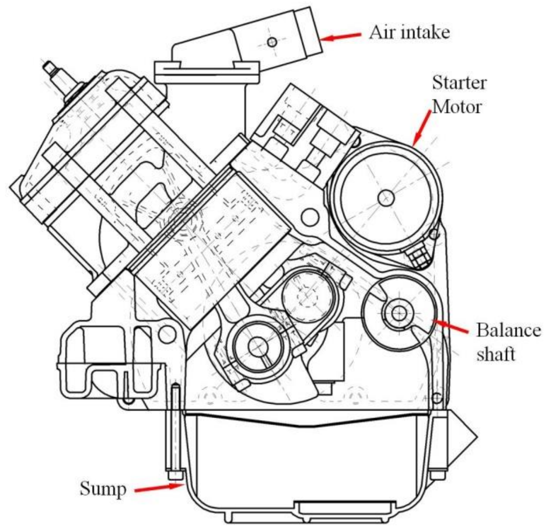

Fig.7. Stepped piston 290 cm³ 180° parallel twin cylinder engine with counter balance shaft
(image courtesy of Bernard Hooper Engineering Ltd)

Table.1. Reference data for engine comparisons

Engine			UMA290 Stepped piston	Four stroke 4S 374
Cycle			2	4
Cylinders			2	2
Swept volume		(cm³)	290	374
Cylinder bore diameter		(mm)	62	62
Stroke		(mm)	48	62
Compression ratio			6.2:1	10.0:1
Dimensions:	length	(mm)	347	
	width	(mm)	325	
	height	(mm)	355	
Engine mass (without starter motor)		(kg)	15.4	

The power plant for a Range Extender EV is an integral part of the overall vehicle forming an essential part of the system whether the engine is operating to improve the state of vehicle battery charge or is in a standby condition. Minimisation of the engine system mass and volume is therefore a key objective within the vehicle architecture and the ultimate ability to maximise efficiency and reduce CO_2 emissions. The small envelope of the UMA290 indicated in Table 1 together with the low basic engine mass of 15.4 kg offer attractive potential to meet these objectives for a small RE-EV.

A significant variable available with a segregated pump charging system is the ability to displace a different swept volume to the volume in the power cylinder. In a conventional crankcase-scavenged two-stroke engine the underside of the piston is used to compress the incoming charge in the crankcase. The air displacement ratio is therefore unity with a single diameter piston. With the UMA290 engine the ratio is 1.1:1. Higher delivery ratios are possible, notwithstanding the consequential increase in cylinder centre distance. The compression ratio of the pumping cylinder system is 1.5:1. Conventional crankcase-scavenged two-stroke engines typically operate at similar crankcase compression ratios to the pump compression ratio selected for the UMA290. Where highly advanced exhaust systems can be applied, harnessing the gas dynamic charging advantages therein, the crankcase compression ratio is often reduced to levels typically between 1.35 and 1.4:1.

4 COMPUTATIONAL MODELLING

In order to provide a basis for comparison, suitable computational models have been developed for both the stepped piston engine and a four stroke engine of comparable performance output. This work has enabled initial modelling and comparison of the likely emissions output of both fundamental engine types.

4.1 STEPPED PISTON ENGINE MODEL

The two cylinder UMA290 engine has been the subject of an initial computational modelling study using the 1-d CFD code WAVE [20] developed by Ricardo. This work builds upon prior study and WAVE modelling of the 4-cylinder SPV580 unit operating on heavy fuels for military application [16]. The basic structure of the UMA290 engine model can be seen in Fig 8. The prior modelling work on the 4-cylinder unit used a parent child modelling strategy where the pumping cylinder geometry and boundary conditions are computed as the parent model which then supplies the boundary conditions to the crossover and power cylinder or child model. This approach has been used again for the analysis conducted within this study. The components of the model elements are indicated in Fig 8. The communication between the parent and child models is conducted through the respective external CFD junctions.

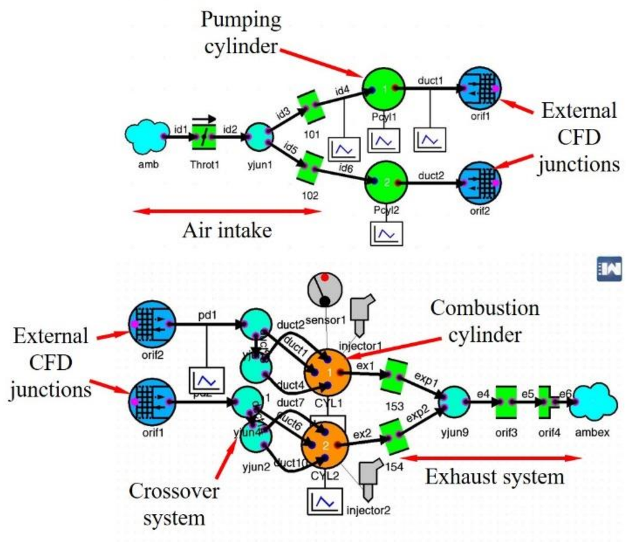

Fig.8. WAVE models of pumping cylinders (parent model) and crossover system / power cylinders (child model) – UMA290 engine

MODEL THEORY

The well-established Wiebe [21] function is employed in order to simulate the combustion process. The function derived within the model was developed based upon the prior work of Heywood and Sher [22] and Sher [23] where analysis of representative Wiebe functions for conventional spark ignition two-stroke cycle engines where researched. The mass fraction burned used within the WAVE models is based upon the function shown in Fig 9.

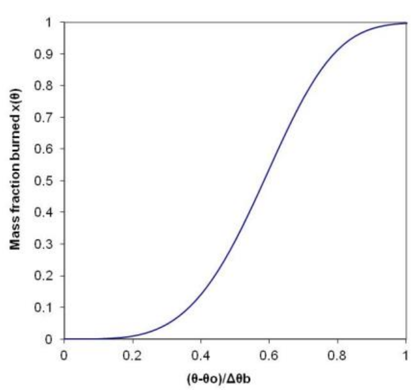

Fig.9. Wiebe function used for the stepped piston engine combustion model

The function defined in Fig 9 is determined from the following expression: -

$$x(\theta) = 1 - exp\left[- \left[\frac{c(\theta - \theta_0)}{\Delta\theta_b} \right]^{-b} \right] \quad (1)$$

Heywood and Sher found that values of *c* from 3 to 3.2 and a value of 5 for *b* could be recommended for small spark ignition two-stroke cycle engines. This was generated from the combustion rate analysis of three crankcase scavenged engines. As a further development Sher and Zeigerson [24] have also modelled a theoretical engine based on the prior work by Sher and Harari [25]. This engine employed stepped pistons for the charge scavenging process. Values for *c* and *b* similar to the values developed by Heywood and Sher have therefore been used to derive the Wiebe function shown in Fig 9 for the UMA290.

In order to represent friction assumptions within the models and attempt to achieve closer correlation between modelled and actual values, Morse test data recorded during prior engine dynamometer testing has been analysed. The Morse tests were performed using a 4-cylinder version of the engine (SPV580) which shares similar core components. Relevant values for mechanical efficiency and friction mean effective pressure (FMEP) recorded at BHE have therefore been used to develop a representative engine friction model with varying speed. This is based upon the well-established friction correlation developed by Chen and Flynn [26] and reproduced for reference in equation (2): -

$$FMEP = C_1 + C_2(P_{max}) + C_3\left(RPM \times \frac{S}{2}\right) + C_4\left(RPM \times \frac{S}{2}\right)^2 \quad (2)$$

The calibration constants C_1 to C_4 are developed as factors to influence the maximum cylinder pressure (P_{max}) and mean piston velocity elements of equation (2).

The original intention of the performance of Morse tests was not to directly assess FMEP but was actually undertaken to make an assessment of engine mechanical efficiency. This is an effective measure of the contribution of each cylinder to the total engine power output; however, this provided useful data for influence of the Chen and Flynn model albeit only over a limited range of engine speeds.

In-cylinder heat transfer assumptions for the WAVE models are based upon the correlations developed by Woschni [27] based upon his experiments exploring firing and non-firing (externally motored) engines. The Woschni derived heat transfer coefficient can be determined from equation (3).

$$h = 0.0128\, D^{-0.2}\, P^{0.8}\, T^{-0.53}\, v_{ch}^{\ 0.8} C_m \quad (3)$$

The characteristic gas velocity v_{ch}, in equation (3) can be defined as a function of the mean piston velocity. This velocity varies as a function of the variation of cylinder pressure. The variable C_m, in equation (3) is a heat transfer area scaling factor defined relative to cylinder bore area. Relevant

C_m values allowing for the increase in surface area relative to a flat top piston (unity) value were therefore input to allow for the actual relevant combustion chamber and piston crown surface areas. Heat transfer coefficients, elsewhere within the models for the circular duct geometry definitions, can be determined from the following expression developed by Colburn [28]: -

$$h = \frac{C_f}{2} \times \rho U c_p Pr^{-\frac{2}{3}} \tag{4}$$

The friction coefficient, C_f, is calculated based upon the laminar or turbulent flow regime occurring within the model.

4.2 FOUR STROKE ENGINE MODEL

As a comparator for the UMA290 stepped piston engine a four stroke engine model was also developed of a suitable swept volume to provide a similar performance output within the same Ricardo WAVE software suite. The engine has a swept volume of 374 cm^3 and the basic model structure can be seen in Fig 10.

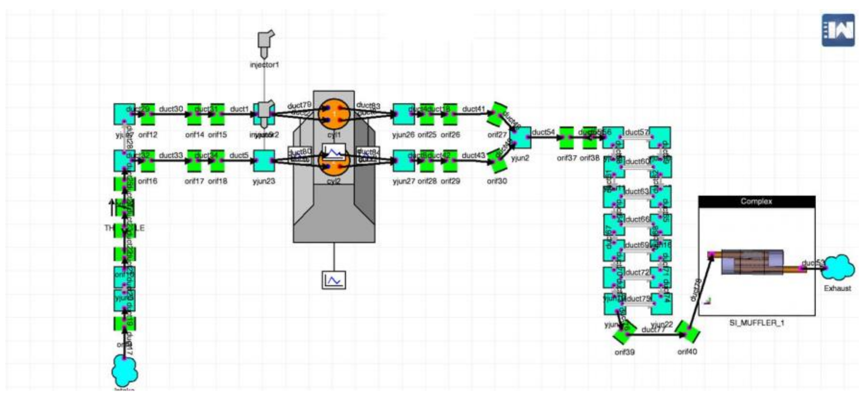

Fig.10. WAVE model of 374 cm^3 four stroke parallel twin cylinder engine

The four stroke engine model uses the parameters listed in Table 1 and in terms of combustion models, friction correlation, etc. the model is based on Ricardo recommendations in order to provide a realistic comparison.

4.3 OPERATING FUELS

Both the stepped piston two stroke and conventional four stroke models were run using indolene as the operating fuel. Indolene is a reference grade form of gasoline definable within the WAVE environment. Octane values for fuels can be defined within the knock model present in WAVE and this was set to 95RON for the selected indolene fuel. The experimental data presented for the UMA290 engine was recorded using 95RON gasoline.

5 1-d CFD MODEL RESULTS

During the experimental research and development phases of dynamometer testing of the UMA290 and SPV580 engines it was not necessary to instrument the test engines with pressure transducers. There is therefore no available data to correlate the pressures occurring within the experimental units with the data predicted by the WAVE models. During development of a precursor engine at Norton Villiers in the 1970s, a 500 cm^3 two-cylinder parallel twin designated SPX500 was developed for the WULF motorcycle as reported by Hooper and Favill [19]. This engine was instrumented with transducers allowing capture of variation of dynamic pressure occurring within the crossover transfer passage. More recently data has been recorded from a compression ignition diesel V4 cylinder stepped piston engine of 1775cm^3 displacement. The data recorded from these experimental engines is presented in Fig 11 for comparative purposes with the WAVE model output for the UMA290.

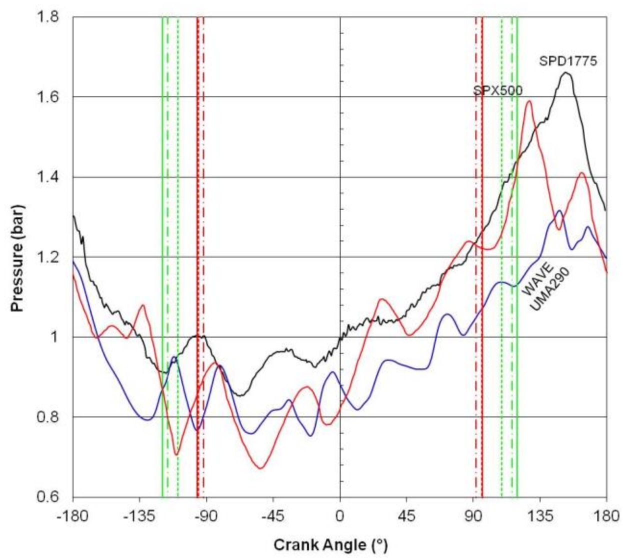

Fig.11. Comparison of stepped piston pumping cylinder pressure for SPX500 and WAVE UMA290 two cylinder engines and SPD1775 stepped piston V4 diesel engine

Data shown in Fig 11 for the SPX500 engine and WAVE UMA290 model is recorded at 5000 rpm. The data reproduced for the compression ignition diesel SPD1775 engine was recorded at 3000 rpm. Port timings are superimposed onto Fig 11 for reference with the SPX500 timings shown as chain dash, the SPD1775 timings are shown as dashed and the UMA290 shown as solid lines. Reasonable correlation in terms of profile shape can be observed but the peak pressure of 1.3 bar occurring at 148° is noticeably lower for the UMA290.

5.1 COMPARISON OF EXPERIMENTAL AND THEORETICAL PERFORMANCE

EXPERIMENTAL PERFORMANCE

The UMA290 engine has been tested experimentally using 95RON gasoline injected into the inlet tract together with similar testing using carburettor fuelling. This data has provided a useful bench mark for comparison of model predicted data with actual experimental results. All experimental data is corrected for standard atmospheric conditions to SAE Standard J1349 [29].

1-d CFD MODEL PERFORMANCE

The ability to model the engines within the environment of WAVE has enabled exploratory work to be conducted using more advanced fuelling methods including direct injection (DI). Such systems have not been available for experimental testing however the potential benefits of DI compared with inlet injection (II) have been explored. Application of DI would be a necessity to minimise emissions for any engine operating on the two stroke cycle for a Range Extender or Hybrid Electric Vehicle. In order to represent a simple DI fuel injection system a relatively low fuel pressure of 5 bar has been modelled which minimises the parasitic losses imposed by the fuel system. This fuel pressure level is similar in magnitude to that of other researchers [8][14], however it is not possible at this stage to consider the effects of air assistance as part of the fuel delivery system. The four stroke engine model could be considered to be able to be emissions compliant with multi point injection just before the inlet valves.

Comparisons of full load performance for both modelled engines using indolene fuel is compared in Figs 12 and 13 with experimental data recorded from the UMA290 engine using gasoline.

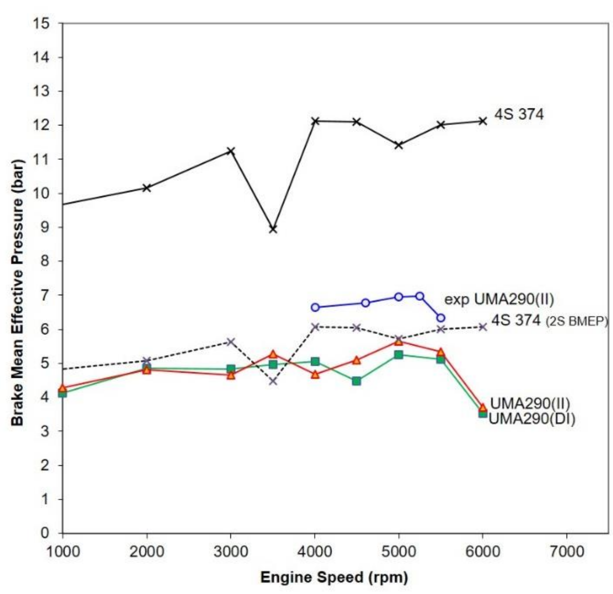

Fig.12. Comparison of BMEP for UMA290 290cm^3 stepped piston engine with inlet injection from experimental data (exp) and WAVE modelled inlet injection (II) and direct injection (DI) versions and an equivalent-performance 374cm^3 four stroke engine (4S 374)

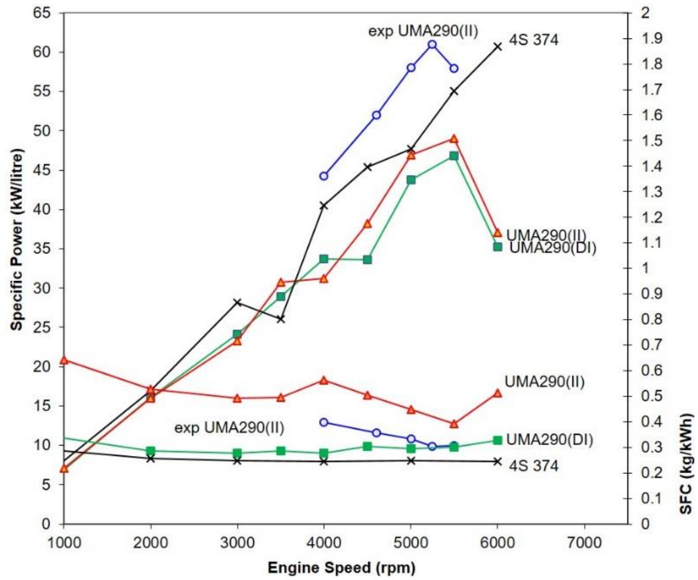

Fig.13. Comparison of specific performance for UMA290 290cm^3 stepped piston engine with inlet injection from experimental data (exp) and WAVE modelled inlet injection (II) and direct injection (DI) versions and an equivalent-performance 374cm^3 four stroke engine (4S 374)

Computational model data shown in Figs 12 and 13 uses indolene as a fuel. Experimental data was recorded using 95RON gasoline. The maximum brake mean effective pressure recorded during experimental testing was 6.97 bar. From the WAVE the corresponding maximum BMEP values can be observed in Fig 12 at 5000 rpm for both inlet injection (II) and direct injection (DI) with values of 5.6 bar and 5.3 bar respectively. The four stroke 374cm^3 twin cylinder engine exhibits a maximum BMEP of 12.1 bar at 4000 rpm. For comparison purposes an equivalent two stroke BMEP is shown as a dotted line in Fig 12 so that a cross reference can be made between the engine types.

The maximum power from experimental testing occurs at 5250 rpm equating to a specific power output of 61 kW/litre. The maximum power observed with WAVE models operating with inlet injection (II) and direct injection (DI) occurs at 5500 rpm with levels of 49 kW/litre and 47 kW/litre respectively. The 374cm^3 four stroke twin cylinder engine can be seen to be predicted to produce 60.7 kW/litre at 6000 rpm.

In terms of specific fuel consumption, the experimental data lies between the WAVE II and DI model predictions. Minimum SFC from the inlet injected model is 0.392 kg/kWh at 5500 rpm. The 374cm^3 four stroke twin cylinder engine can be seen to be predicted to produce a minimum SFC of 0.244 kg/kWh at 6000 rpm. Using direct injection, the model prediction shows a minimum SFC of 0.278 kg/kWh occurring at both 3000 and 4000 rpm for the UMA290 engine.

5.2 EMISSIONS MODELLING

WAVE modelling of full load emissions is compared for the UMA290 using inlet injection (II) and direct injection (DI) with the 374cm^3 four stroke twin cylinder engine using port injection in

Figs 14 to 17. In all cases the engines are operating at stoichiometric air:fuel ratios. Models for both DI and II assess the air flow entering the cylinder and match the fuel delivery to the monitored air mass flow rate. Stratified operation has shown good results on experimental stepped piston charged engines but no lean operation has been considered during this phase of research. DI operation at this stage of the study was maintained at a low line operating pressure as discussed earlier however variable injection timing was explored to capture the summarised data presented.

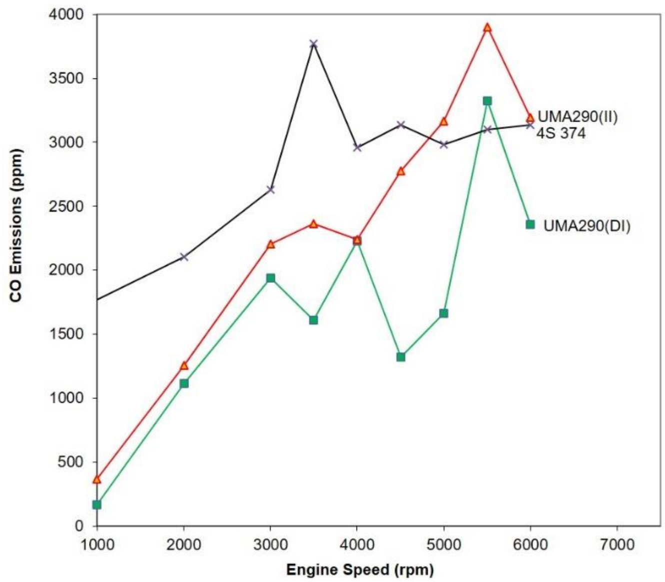

Fig.14. Comparison of predicted CO emissions for UMA290 290cm^3 stepped piston engine with WAVE modelled inlet injection (II) and direct injection (DI) versions and an equivalent-performance 374cm^3 four stroke engine (4S 374)

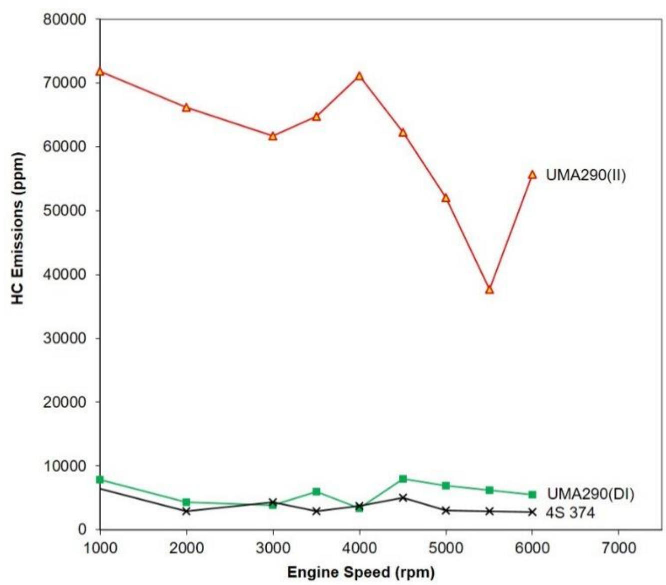

Fig.15. Comparison of predicted HC emissions for UMA290 290cm^3 stepped piston engine with WAVE modelled inlet injection (II) and direct injection (DI) versions and an equivalent-performance 374cm^3 four stroke engine (4S 374)

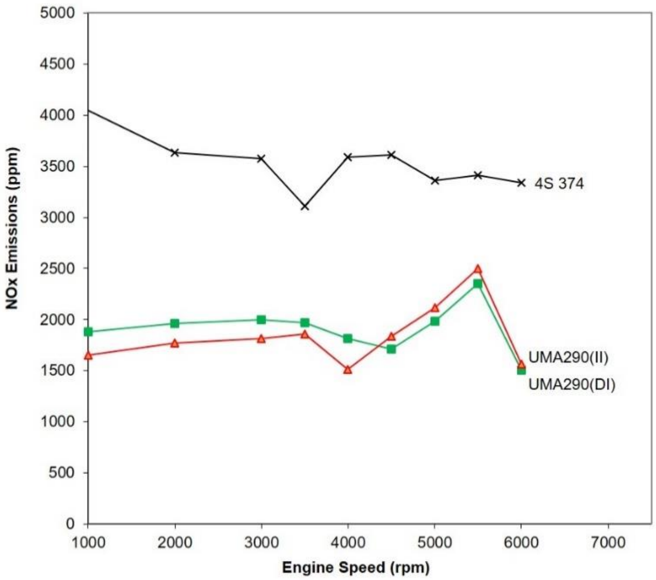

Fig.16. Comparison of predicted NOx emissions for UMA290 290cm^3 stepped piston engine with WAVE modelled inlet injection (II) and direct injection (DI) versions and an equivalent-performance 374cm^3 four stroke engine (4S 374)

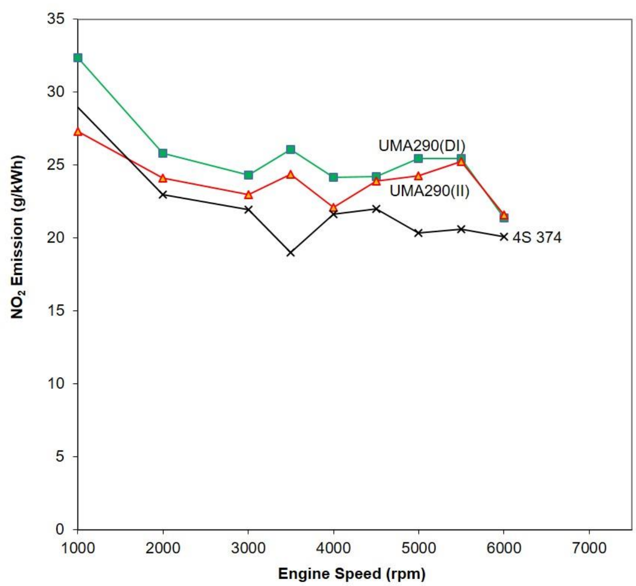

Fig.17. Comparison of predicted NO_2 emissions for UMA290 290cm^3 stepped piston engine with WAVE modelled inlet injection (II) and direct injection (DI) versions and an equivalent-performance 374cm^3 four stroke engine (4S 374)

From Fig 14 it can be observed that the direct injected UMA290 model generally emits the lowest overall CO emission with the exception being at 5500 rpm where the 374cm^3 four stroke engine emits a marginally lower level. The lowest overall CO emission from the inlet injected UMA290 model is 368 ppm at 1000 rpm rising to 3900 ppm at 5500 rpm. The direct injection version of the same engine exhibits lower CO emission at the same speeds of 164 ppm at 1000 rpm rising to 3326 ppm at 5500 rpm. The four stroke engine by comparison emits 1767 ppm at 1000 rpm rising to 3136 ppm at 6000 rpm with a maximum level of 3769 ppm occurring at 3500 rpm. The DI fuel pressure has been limited in this study to 5 bar which may be limiting effective vapourisation and charge mixing. It should also be pointed out that the port timing for the UMA290 engine is optimised to achieve maximum power in the 5000 – 5500 rpm speed range. The port timing is therefore not ideal for operation at lower speeds. The addition of a charge trapping device similar to the systems applied by Mattarelli *et al* [6] or Turner *et al* [10] could assist with performance enhancement and emissions reduction at lower operating speeds.

Unburnt hydrocarbon emission levels shown in Fig 15 are highest for the inlet injected UMA290. The application of DI shows significant reduction in HC emission with a minimum emission level occurring at 4000 rpm. The 374cm^3 four stroke twin cylinder engine emits the lowest HC emission value with the minimum level being 2850 ppm at 6000 rpm; however the DI UMA290 shows lower HC emission at the model speeds of 3000 rpm and 4000 rpm with predicted values of 3811 ppm and 3413 ppm respectively. Charge short circuiting is possible with all fuelling configurations but particularly so with an inlet injected engine where the fuel air charge enters the cylinder pre-mixed with incoming air charge around bottom dead centre whilst the exhaust port is also open. The significant reduction in HC emission via application of direct injection could be expected to be a result of the reduced possibility of fuel short circuiting as the fuel is injected later in the cycle. The levels are however still higher than the comparable four stroke engine.

Oxides of nitrogen present one of the most difficult emissions to control and convert. The benefit of the lower combustion temperature and its effect on reducing NOx emission is clearly evident with the profiles shown in Fig 16 for the two stroke cycle UMA290. The lowest NOx emission level using inlet injection occurs at 4000 rpm with a minimum value of 1511 ppm. Applying DI similarly shows a reduction in NOx emission level from 4500 rpm to 6000 rpm with predicted minima of 1714 ppm and 1505 ppm at 4500 rpm and 6000 rpm respectively. In comparison the 374cm^3 four stroke twin cylinder engine emits a minimum NOx level of 3110 ppm at 3500 rpm.

Observed NO_2 emissions predicted from the WAVE models are presented in Fig 17. The 374cm^3 four stroke engine generally exhibits the lowest overall NO_2 emission apart from at 1000 rpm with the minimum NO_2 emission of 19.0 g/kWh occurring at 3500 rpm. The lowest values for the UMA290 are observed using inlet injection with levels of 22.1 g/kWh occurring at 4000 rpm and 21.6 g/kWh at 6000 rpm. At the same speeds the DI version of the UMA290 shows minimum values of 24.2 g/kWh at 4000 rpm (also occurring at 3000 and 4500 rpm) and 21.4 g/kWh at 6000 rpm.

6 DISCUSSION

The viability of production of small compact Range Extender or Hybrid Electric Vehicles relies heavily on the need for an efficient low emission IC engine power plant system with low production cost characteristics. A limit to the number of cylinders can be imposed to minimise cost as demonstrated by the work of Bassett *et al* [4] or Mattarelli *et al* [2][6] where parallel twin cylinder units have been considered. Horizontally opposed flat twin engines have also been considered for HEV application as demonstrated by the Peugeot Citroen Dynavolt system and the research published by Stan and Personnaz [5].

The ability to achieve significantly lower levels of NOx emission is a key advantage of the two stroke cycle engine. The work presented here has demonstrated potential for a power plant solution offering low emissions whilst retaining a low production cost base. The stepped piston engine has demonstrated low NOx emission for a motorcycle operating on the LA4 test cycle as reported by Hooper *et al* [12]. The low NOx emissions modelled in this study offer significant potential to contain this most challenging emission. Operation of a Range Extender engine at idle operating conditions may be limited depending on the final application as the power plant would most likely be shut down at anything other than the most efficient operating condition of the engine. However if operation at idle is a requirement the prior work demonstrating extremely low NOx emission reported by Blundell *et al* [11] as applied to the Lotus Omnivore two stroke engine could equally be applied to a stepped piston charged engine with the addition of variable compression ratio technology.

Modelling of potential raw carbon monoxide and unburnt hydrocarbon emissions has also been possible with the assistance of the WAVE models and in particular the ability to model the effects of applying direct injection. Results from comparative CO and HC emissions modelling for the UMA290 and the comparable theoretical 374 cm^3 four stroke engine unit so far exhibit lower CO and HC emission for the four stroke engine. DI would be essential for future low emission engines operating on the two stroke cycle. As previously discussed the durability problems of conventional crankcase scavenged engines as ever increasing efficiency and decreasing exhaust gas emission levels are imposed presents significant challenges is unlikely to diminish.

As previously discussed the durability problems of conventional crankcase scavenged engines presents significant challenges. The problems are compounded for conventional engines by the need for ever increasing efficiency and decreasing exhaust gas emission levels. This means that thermal loading of key components will increase. The piston in particular is the key aspect affected and enhanced cooling will be required to secure low emission targets with acceptable levels of engine endurance in service. Segregating the charging process by application of stepped pistons is one such method that can address this challenge. Furthermore the significant improvements of minimising oil consumption as experienced on several prototype units would appear to be able to address the concerns expressed by Bassett *et al* [4] in relation to conventional two stroke engines. Further work is required in these areas but the 75% improvement compared with conventional engines is a promising starting point for this.

Performance levels for the 290 cm^3 UMA290 compare favourably with the 374 cm^3 four stroke engine with similar specific power output. In terms of specific fuel consumption, the four stroke engine generally shows lower levels. If this is further analysed in an overall vehicle installation context, the UMA290 would present a lower power plant system mass which would erode the SFC superiority of a comparable vehicle with the 374 cm^3 four stroke two-cylinder unit as the IC engine power source.

Two cylinder engines depending on their configuration can exhibit rocking couples. A 180° disposition of the cylinders provides a reasonably compact and relatively straightforward packaging solution for exhaust system installation in the vehicle. However this arrangement would present such a rocking couple. Application of a counter balance shaft as shown in Fig 7 can however eliminate this, albeit at additional cost. The addition would offer beneficial characteristics in terms of refinement from an NVH viewpoint. Installation within a semi anechoic power plant cell could also assist reduction of the well reported negative NVH characteristics of periodic power plant operation which is typical of RE-EVs and HEVs. The unexpected start up of the IC engine unit often provides an unexpected and unwanted noise source within the vehicle structure and destroys the near silent driving experience as discussed by Millo *et al* [30]. Application of a partially insulated noise cell could minimise this negative vehicle characteristic in RE-EVs and HEVs offering the potential for low emission high efficiency transportation in a low cost small vehicle.

7 CONCLUSIONS

The study reported within this paper presents results of work conducted to date. Further work is required but the potential for an alternative low emission IC engine power plant system suitable for small RE-EVs and HEVs offering the benefits of two stroke cycle operation with high specific performance from a compact low mass unit is a distinct possibility. Specific power output of 61 kW per litre has so far been demonstrated. Application of direct injection has predicted a minimum full load SFC using indolene fuel of 0.278 kg/kWh at 3000 rpm and 4000 rpm. Corresponding minimum full load NOx emission is between 45 to 69% of the comparative four stroke engine model output level. The two stroke operating cycle also has the advantage of improved NVH characteristics for the same disposition of cylinders which is a critical area requiring attention in future compact low mass HEVs and/or RE-EVs where two cylinder units may be the likely IC engine choice.

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APPENDIX

Notation

		Units
AFR	air:fuel ratio	
b	constant (form factor)	
c	constant (efficiency parameter)	
C_1	constant	
C_2	constant	
C_3	constant	
C_4	constant	
C_f	friction coefficient	
C_m	relative heat transfer area scaling factor	
c_p	gas constant pressure specific heat	(J/kgK)
D	cylinder diameter	(m)
$FMEP$	friction mean effective pressure	(bar)
$IMEP$	indicated mean effective pressure	(bar)
h	heat transfer coefficient	(W/m^{2}K)
P	instantaneous gas pressure	(bar)
P_{max}	maximum cylinder pressure	(bar)
Pr	Prandtl number	
RPM	engine speed	(r/min)
S	engine stroke	(m)
T	instantaneous gas temperature	(K)

U	gas velocity	(m/s)
v_{ch}	characteristic gas velocity	(m/s)
$x(\theta)$	mass fraction burned at crank angle θ	(°)
$\Delta\theta_b$	duration of combustion	(°)
θ	crank angle	(°)
θ_0	crank angle at start of combustion	(°)
ρ	gas density	(kg/m^3)

Abbreviations

BDC	Bottom Dead Centre
BMEP	Brake Mean Effective Pressure
1-d CFD	One dimensional Computational Fluid Dynamics
CI	Compression Ignition
CO	Carbon Monoxide
DI	Direct Injection
DOHC	Double Overhead Camshaft
EC	Exhaust port Closure
EO	Exhaust port Opening
FMEP	Friction Mean Effective Pressure
GM	General Motors
HC	unburnt Hydrocarbons
HEV	Hybrid Electric Vehicle
IC	Internal Combustion
II	Inlet Injection
IMEP	Indicated Mean Effective Pressure
NOx	Oxides of Nitrogen
NVH	Noise Vibration and Harshness
PPM	Parts Per Million
RE-EV	Range Extender Electric Vehicle
RON	Research Octane Number
SFC	Specific Fuel Consumption
SI	Spark Ignition
SPD1775	Stepped Piston Diesel V4 cylinder 1775 cm^3 engine
SPV580	Stepped Piston V4 cylinder 580 cm^3 engine
SPX	Stepped Piston crossover system
SPX500	Stepped Piston twin-cylinder 497cm^3 engine developed at Norton Villiers/BHE
TC	Transfer port Closure
TDC	Top Dead Centre
TO	Transfer port Opening
UAV	Unmanned Air/Aerial Vehicle
UMA290	Unmanned Aircraft Stepped piston 290 cm^3 twin-cylinder engine
VCR	Variable Compression ratio